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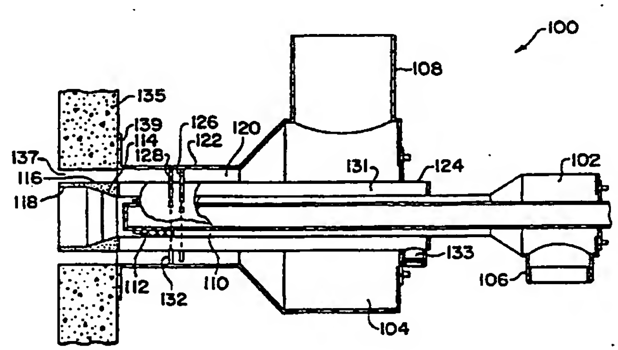

(57) Abstract

A forced draft burner apparatus for burning a gaseous fuel while producing low levels of NO_x, CO and hydrocarbon emissions comprising: a cylindrical inner burner having a tubular wall; a generally cylindrical body mounted inside the tubular wall of the inner burner; an annular flow channel (110) being defined between said body and the inner wall of said tubular section, said channel constituting a throat for oxidant gases, and having a downstream outlet for the inner burner; means (102, 106) for supplying oxidant gases to said throat of the inner burner; a divergent quarl (116) for said inner burner having its smaller end connected to said outlet of said inner burner, and exiting into a combustion chamber; a plurality of curved axial swirl vanes (112) being mounted in said annular flow channel of the inner burner to impart swirl to said oxidant gases flowing downstream in said throat; inner burner fuel gas injection means for the inner burner being provided in said annular channel proximate to said swirl vanes for injecting said gas into the flow of oxidant gases at a point upstream of said outlet end; an outer burner surrounding at least a portion of said inner burner and including a wall spaced from the outer wall of the inner burner to define an outer burner flow channel (120) having a downstream outlet end for gases provided to said channel; means for providing a flow of oxidant into the outer burner flow channel (104, 108); and outer burner fuel gas injection means (126, 128) for the outer burner being provided in said outer burner flow channel, upstream of the outer burner outlet end.

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APPARATUS AND METHOD FOR REDUCING NO_x, CO AND HYDROCARBON EMISSIONS WHEN BURNING GASEOUS FUELS

FIELD OF THE INVENTION

This invention relates generally to combustion apparatus, and more specifically relates to a burner that combines the advantageous operating characteristics of nozzle mix and premixed type burners to achieve extremely low NO_x, CO and hydrocarbon emissions.

BACKGROUND OF THE INVENTION

NO_x emissions from gas flames can be created either through the Zeldevitch mechanism (often called thermal NO_x) or through the formation of HCN and/or NH_3 which can then be ultimately oxidized to NO_x (prompt NO_x). Thermodynamic calculations typically show that NO_x emissions measured from natural gas flames are well below, one to two orders of magnitude, the thermodynamic equilibrium value. This indicates that in most situations NO_x formation is kinetically controlled.

Kinetic calculations indicate that thermal NO_x emissions are typically the most important source of NO_x for natural gas flames, with the NO_x being created through the following reactions:

$$N + O_2 = NO + O \qquad (1)$$
$$N + OH = NO + H \qquad (2)$$
$$N_2 + O = NO + N \qquad (3)$$

Kinetic calculations were performed using a PC version of the CHEMKIN computer program. Calculations using this program have provided valuable insight into changes in the burner fuel and air mixing characteristics which can lower NO_x emissions.

As the name implies, thermal NO_x can be controlled by regulation of the peak flame temperature, and as shown in Figure 1 using kinetic calculations, if the temperature can be lowered enough the NO_x emissions from a "true" premixed natural gas flame operating at 15% excess air can be reduced to extremely low values (less than 1 ppmv). In effect Figure 1 shows the relationship between thermal NO_x and temperature since for a premixed natural gas flame with an excess of oxygen, thermal NO_x is the only route by which any significant NO_x emissions are created.

Under appropriate flame conditions the formation of prompt NO_x can also be important when burning natural gas. The kinetic model used shows that under fuel rich conditions, particularly when the stoichiometry is under about 0.6, both HCN and NH_3 can be formed through reaction of CH with N_2 to form HCN and N. These calculations were conducted using gas and air mixtures with stoichiometries ranging from 1.0 to 0.4. The model predicts that prompt NO_x becomes important at higher stoichiometries when the temperature is lower; see Figure 2. Below a stoichiometry of 0.5 almost all the NO_x formed is prompt NO_x. The rate of prompt NO_x formation (as the name implies) is also very rapid, being nearly complete in about 1 millisecond at a temperature of 2400° F.

Kinetic calculations also indicate that hydrocarbon fragments, in addition to being important for prompt NO_x, are also important for thermal NO_x formation since they can act as a source of O atoms and OH radicals. Kinetic calculations show the importance of the hydrocarbon concentration in the formation of NO_x, even under oxidizing conditions. At a temperature of 3400° F the predicted NO_x emissions were about 4 ppmv after 5 ms residence time for a mixture of N_2, O_2, H_2O, and CO_2 when hydrocarbons were not present, as compared to 80 ppmv

when combustion of about 1% CH_4 was present in the gas mixture. If the concentration of methane initially present was reduced to about 0.5%, the NO_x concentration after 5 ms was reduced to about 75 ppmv. The kinetic model used predicts that the following mechanisms are important:

1. Reaction of CH_4 with O_2, OH and H to form CH_3
2. Reaction of CH_3 with O_2 to form CH_3O and O
3. Reaction of N_2 with O to form NO and O
4. Various reactions to form OH
5. Reaction of N_2 with OH to form NO and NH

Low NO_x gas burners have been undergoing considerable development in recent years as governmental regulations have required burner manufacturers to comply with lower and lower NO_x limits. Most of the existing low NO_x gas burner designs are nozzle mix designs. In this approach the fuel is mixed with the air immediately downstream of the burner throat. These designs attempt to reduce NO_x emissions by delaying the fuel and air mixing through some form of either air staging or fuel staging combined with flue gas recirculation ("FGR"). Delayed mixing can be effective in reducing both flame temperature and oxygen availability and consequently in providing a degree of thermal NO_x control. However, delayed mixing burners are not effective in reducing prompt NO_x emissions and can actually exacerbate prompt NO_x emissions. Delayed mixing burners can also lead to increased emissions of CO and total hydrocarbons. Stability problems often exist with delayed mixing burners which limit the amount of FGR which can be injected into the flame zone. Typical FGR levels at which current burners operate are at a ratio of around 20% recirculated flue gas relative to the total stack gas flow.

A further type of low NO_x burner which has been developed in recent years is the premixed type burner. In this approach, the fuel gas and oxidant gases are mixed well upstream of the burner throat, e.g. at or prior to the windbox. These burners can be effective in reducing both thermal and prompt NO_x emissions. However, problems with premixed type burners include difficulty in applying high air preheat, concerns about flashback and explosions, and difficulties in applying the concept to duel fuel burners. Premix burners also typically have stability problems at high FGR rates.

In the inventions of my S.N. 092;979 and 188,586 applications (the disclosures of which are hereby incorporated by reference) extremely low NO_x, CO and hydrocarbon emissions are achieved, while maintaining the desirable features of a nozzle mix burner. This is accomplished by injecting the fuel gas, such as natural gas, in a position that would be typical for a nozzle mix burner, while generating such rapid mixing that, effectively, premixed conditions are created upstream of the ignition point.

In such burner apparatus an outer shell is provided which includes a windbox and a constricted tubular section in fluid communication therewith. A generally cylindrical body is mounted in the shell, coaxially with and spaced inwardly from the tubular section so that an annular flow channel or throat is defined between the body and the inner wall of the tubular section. Oxidant gases are flowed under pressure from the windbox to the throat, and exit from a downstream outlet end. A divergent quarl is adjoined to the outlet end of the throat and define a combustion zone for the burner. A plurality of curved axial swirl vanes are mounted in the annular flow channel to impart swirl to the oxidant gases flowing downstream in the throat. Fuel gas injector means are provided in

the annular flow channel proximate or contiguous to the swirl vanes for injecting the fuel gas into the flow of oxidant gases at a point upstream of the outlet end. The fuel gas injection means comprise a plurality of spaced gas injectors, each being defined by a gas ejection hole and means to feed the gas thereto. The ratio of the number of gas ejection holes to the projected (i.e. transverse cross-sectional) area of the annular flow channel which is fed fuel gas by the injector means is at least $200/ft^2$.

One or more turbulence enhancing means may optionally be mounted in the throat at at least one of the upstream or downstream sides of the swirl vanes. These serve to induce fine scale turbulence into the flow to promote microscale mixing of the oxidant and fuel gases prior to combustion at the quarl.

The gas injectors can be located at the leading or trailing edges of the swirl vanes, and inject the fuel gas in the direction of the tangential component of the flow imparted by the swirl vanes. The gas injectors can also be disposed on a plurality of hollow concentric rings which are mounted in the throat downstream of the swirl vanes. The injectors can similarly comprise openings disposed in opposed concentric bands on the walls which define the inner and outer radii of the annular flow channel. The gas injectors can also be located at the surfaces of the swirl vanes, with the vanes being hollow structures fed by a suitable manifold. Preferably the geometry of the burner is such that the product of the swirl number S and the quarl outlet to inlet diameter ratio C/B is in the range of 1.0 to 3.0.

Pursuant to another aspect of the S.N. 092,979 and S.N. 188,586 invention, a method is provided for injection of gaseous fuel in a forced draft burner of the type which

includes an annular throat of outer diameter B, having an inlet connected to receive a forced flow of air and recirculated flue gases, and an outlet adjoined to a divergent quarl. The gaseous fuel is injected at an axial coordinate which is spaced less than B in the upstream direction from the axial coordinate at which the quarl divergence begins; and sufficient mixing of the gaseous fuel with the air and recirculated flue gases is provided that these components are well-mixed down to a molecular scale at the axial coordinate of ignition. This procedure results in extremely low NO_x, CO and hydrocarbon emissions from the burner.

In a further aspect of the S.N. 092,979 and S.N. 188,586 invention, the swirl vanes, which are mounted with their leading edges parallel to the axial flow of fuel and oxidant gases, and then slowly curve to the final desired angle, have a constant radius of curvature along the curved portion of the vane, whereby the curved portion is a section of a cylinder. This shape simplifies manufacturing using conventional metal fabricating techniques.

Additional background which will be helpful in understanding the present invention can be gained by reviewing Figures 3, 4, 5 and 6 herein, which describe a representative embodiment of the apparatus disclosed in my prior applications. In Figure 3 an isometric perspective view thus appears of such prior art embodiment of burner apparatus 51. This Figure may be considered simultaneously with Figures 4, 5 and 6, which are respectively longitudinal cross-sectional; and front and rear end views of apparatus 51.

In burner apparatus 51 combustion air (which can be mixed with recirculated flue gas) is provided to the windbox 53 through a cylindrical conduit 55. Windbox 53 adjoins a

tubular section 57 which terminates at a flange 59, which is secured to a divergent quarl 58 (Figure 12). In the arrangement shown, the inner co-axial cylindrical body 61 is comprised of a central hollow cylindrical tube 63 intended for receipt of an oil gun or a sight glass, and a surrounding tubular member or cylinder 65 which is spaced from the outside wall of tube 63 and closed at each end, by closures 67. A hollow annular space 68 is thereby formed between tubular member 63 and cylinder 65, which serves as a manifold 68 for the fuel gas which is provided to such space via connector 69. The cylindrical body 61 is positioned and spaced within wind box 53 and tubular section 61 by passing through flanges, one of which is seen at 71. The latter is secured to a plate 73 at the end of the wind box by bolts 75 and suitable fasteners (not shown). This arrangement enables easy disassembly, as for servicing and the like.

In the arrangement of burner 51, a series of swirl vanes 77 are provided in the annular space or throat 79 which is defined between tubular body 61 (specifically, between the outer wall of cylinder 65) and the inner wall of tubular member 57.) At the immediately upstream end of each of the swirl vanes 77, gas injector means are provided which take the form of a plurality of tubes 81, each of which is provided with multiple holes 83. It will be evident that the tubes 81, being hollow members, are in communication at their open one end with the interior of the gas manifold 68 defined within member 65, which therefore serves as a feed source for the fuel gas. The fuel gas is discharged in the direction of the openings 83, so that in each instance fuel is injected into the throat directly at the leading edges of the swirl vanes and in the direction of the tangential component of the flow imparted by the swirl vanes 77. Accordingly, the gas injection also acts to enhance the swirl number of the flow.

Although the invention of my S.N. 092,979 and 188,586 applications (hereinafter at times referred to as the "basic rapid mix burner" or "basic RMB") is extremely effective in achieving the desired results, the basic RMB design results in a burner size that is significantly larger than many existing burners. Although the large burner size is not inherently important to the rapid mix feature, the large burner size is important for creating an extremely stable flame which allows high flue gas recirculation rates to be used without concerns about the flame becoming unstable.

Another limitation of the basic RMB design is that the burner geometry must be kept circular. This is clearly a limitation in a boiler or furnace that use square, rectangular or other shape burners.

When the basic RMB is retrofit into existing furnaces, the larger size, relative to the existing burner, can create significant difficulties and increase the retrofit cost. Problems with the larger burner size are particularly apparent when the boiler or furnace burner wall is a "water wall" consisting of pressurized steam or water tubes. For this type design the burner openings are made by bending the boiler tubes. Any significant increase in the burner size entails bending new tubes to make a larger opening. Utility and large field erected industrial boilers typically have the burners inserted through a water wall.

One method of reducing the burner size is to increase the velocities through the burner. However this method has the disadvantage of increasing the pressure drop through the burner. A higher pressure drop through the burner creates other retrofit difficulties, including replacement of forced draft fans, increased operating

costs associated with the higher fan pressure, structural limitations on the windbox and increased operating costs.

SUMMARY OF INVENTION

Now in accordance with the present invention, a two stage rapid mix burner design provides apparatus and method which both significantly reduces the burner size of a rapid mix burner, and/or the burner pressure drop, while maintaining the rapid mix feature and stability of the basic rapid mix design. The two stage rapid mix burner design of the invention can also be easily altered to fit in non-circular geometries, such as a corner or tangential fired boiler.

The present invention uses a circular basic rapid mix burner (i.e. as in my earlier applications), located internally inside a larger burner which can be non-circular. The inner burner provides the flow of hot gases which stabilizes the outer burner. In effect, the combustion gases produced in the inner burner replace the strong internal recirculation flow generated by the basic RMB as an ignition source for the outer burner flow. The inner burner uses the same type swirler, burner and quarl geometry as the basic RMB burner described in my previous applications and consequently has the desired stability and NO_x, CO and HC performance. The outer portion of the burner uses a rapid mix injection grid and consequently also has the desired NO_x, CO and HC performance. Since the flame stability is provided by the inner burner, swirl vanes or a divergent quarl for the outer portion of the burner are not required.

The inner burner is circular with a cylindrical tube mounted in the center defining an annular space between the outer and inner tubes. A plurality of curved fixed axial vanes are mounted in the annular space to impart

swirl to the oxidant gases flowing through the burner. The number of vanes varies linearly with the burner diameter. The typical spacing between vanes, on the inner annulus is approximately one inch. Fuel injection means are provided in the annular flow channel proximate or contiguous to the swirl vanes for injecting the fuel gas into the flow of oxidant gases. The fuel gas injection means comprises a plurality of spaced gas injectors, each defined by a gas injection hole and a means to feed the gas thereto. The ratio of the number of gas injection holes to the projected area of the annular flow channel which is fed fuel gas by the injector means is at least $200/ft^2$. A divergent quarl is adjoined to the outlet end of the inner burner and defines a combustion zone for the burner. The purpose of the quarl is to both promote strong internal recirculation within the inner burner and to provide enough residence time to allow the stability of the flame from the inner burner to be relatively unaffected by the outer portion of the burner. Consequently, a quarl length/inlet diameter ratio of at least 1.75 is desired.

The gas injectors for the inner burner can be located at the leading or trailing edges of the swirl vanes, and inject the fuel in the direction of the tangential component and/or opposite to the direction of the tangential velocity component of the flow imparted by the swirl vanes. The gas injectors can also be disposed on a plurality of hollow concentric rings which are mounted in the throat downstream of the swirl vanes. The injected gas can similarly comprise openings disposed in opposed concentric bands on the walls which define the inner and outer radii of the annular flow channel. The gas injectors can also be located at the surfaces of the swirl vanes, with the vanes being hollow structures fed by a suitable manifold. Details of these arrangements are shown in my S.N. 092,979 and 188,586 applications.

The inner burner is enclosed by a second annular space or number of outer burners cells for which the inner burner acts as an ignition source. The air to the outer burner annulus or regions can be fed from either a separate windbox or from a windbox common to both the inner and outer burner. The two most common geometries for the outer burner are an annular space concentric to the inner burner or a rectangular region with the inner burner diameter less than or equal to the smaller dimension of the rectangular opening. However the basic two stage RMB concept can function with outer burner geometries of any shape.

Inside the region defined by the oxidant flow of the outer burner, rapid mix gas injectors are positioned to provide rapid mixing between the oxidant and fuel. The gas injectors can take the shape of radial spuds fed from either a outer or inner manifold. The spuds are drilled with holes to provide the desired mixing rate between fuel and oxidant. The gas injection spuds can also take the shape of concentric rings, horizontal or vertical grids or other shapes compatible with the outer burner geometry. Typically the spacing between gas injection spuds is approximately one inch with the spacing between the holes drilled into the spuds being in the range 0.2 to 0.4 inches. The spacing of the fuel gas injection holes provides uniform gas distribution within the oxidant. The cross-sectional area of each gas spud is at least 3 times the total area of the injection holes in each spud, to provide adequate gas distribution to each hole. Typically, if the number of holes in each spud is greater than 4, 1/4 inch diameter cylindrical tubing is preferably not used for the injection spuds. Instead either "racetrack" oval tubing, airfoil tubing or fabricated injectors having a maximum width, in a plane defined by the cross-sectional area of the burner throat, of 1/4 inch and a length, normal to the same plane,

determined by the required cross-sectional area and wall thickness of the tube. The "flattened" faces of these tubes are thus the surfaces at which the ejector holes are present, and thus the direction of gas ejection is generally tangential to a radius drawn to the hole, and in a plane or planes transverse to the axis of the burner.

As an example of a typical injector design, an injection spud may have a height of 3 inches with an average hole spacing of 0.25 inch (resulting in 12 holes). Using 1/16 inch holes, the total injection area would be 0.0368 square inches per spud. If tubing with a 0.035 inch wall is used, and the tube minor axis is 0.25 inch, a length for the major axis of the tube of at least 0.625 inches would be required to maintain a inlet area for the spud of at least 3 times the injection area.

The ratio of the number of gas injection holes to the projected cross-sectional area of the annular flow channel is at least $200/ft^2$. The diameter of the holes is determined by the same criteria as discussed in my prior pending applications.

Means may be provided to enhance the mixing of the gas and oxidant in the outer portion of the burner. These means may include the use of screens or perforated plates which induce fine scale turbulence into the flow, or axial swirl vanes may be used in the outer flow to both induce mixing and to control the flame shape.

The heat input ratio between the inner and outer burners is typically in the range of 5% to 20% when the burner is operated at maximum capacity. In one mode of operation the heat input to the inner portion of the burner would remain fixed and, if a lower heat input is required, the fuel and oxidant rate would be decreased in the outer

burner only. In the extreme case the burner could be operated with fuel input to the inner portion of the burner only, in which case the burner would operate as a standard RMB. However, if desired, the thermal inputs of the inner and outer burner could be controlled together. In this mode the inner and outer burner would be controlled so that the heat input from both burner portions would vary linearly; i.e. if the total input is 50% both the inner and outer burners would operate at 50% of maximum input.

Typically, recirculated flue gas (FGR) is added to the combustion air of both the inner and outer burner. The FGR is added far enough upstream of the burner to result in premixed air and FGR at the gas injection point. As an alternative to FGR, air or another inert can be used to reduce the flame temperature. The amount of FGR used is dependent on the desired NO_x level.

As also disclosed in my said 092,979 and 188,586 applications, an oil gun can be inserted through the center, along the axis of the inner burner, to provide backup oil burning capability. When operated on oil, the swirl vanes and quarl of the inner burner will provide the necessary flame stability. All the oil will be injected through the center of the burner, providing the delayed fuel and air mixing (internal staging) necessary for NO_x control with oils which contain a significant amount of fuel nitrogen.

BRIEF DESCRIPTION OF DRAWINGS

The invention is diagrammatically illustrated, by way of example, in the drawings appended hereto in which:

FIGURE 1 is a graphical depiction showing calculated NO_x versus adiabatic flame temperature for a premixed flame with 15% excess air;

FIGURE 2 is a further graph showing kinetic calculation of prompt NO_x (HCN and NH_3);

In FIGURE 3 a perspective view appears of an embodiment of prior art burner apparatus in accordance with the disclosure of my S.N. 092,979 and 188,586 applications;

FIGURE 4 is a longitudinal cross-sectional view through the apparatus of Figure 3;

FIGURES 5 and 6 are respectively front and rear-end views of the apparatus of Figures 3 and 4.

FIGURE 7 is a longitudinal cross-sectional view, through a first embodiment of apparatus in accordance with the present invention;

FIGURE 8 is a front end view of the Figure 7 apparatus;

FIGURE 9 is a longitudinal cross-sectional view, through a second embodiment of apparatus in accordance with the present invention;

FIGURE 10 is a front end view of the Figure 9 apparatus;

FIGURE 11 is a schematic longitudinal cross-sectional view of a two stage apparatus in accordance with the invention, which is provided with a rectangular outer burner portion;

FIGURE 12 is an end view of the Figure 11 apparatus;

FIGURE 13 is a schematic end view of 6 burners as they would appear in one corner of a typical corner fired burner application;

FIGURE 14 is a graphical depiction showing the effect of varying the ratio of the inner/total burner heat input as a function of FGR rate with ambient air, and also compares the performance of the rectangular two stage RMB with the basic RMB;

FIGURE 15 is a graph showing the CO and total hydrocarbon emissions (THC) as a function of the FGR rate for the two stage burner of the invention;

FIGURE 16 is a graph comparing the effect of the inner/total burner heat input as a function of FGR rate for 500°F air preheat;

FIGURE 17 is a graph illustrating an example of NO_x, CO, and THC performance of the invention as a function of excess air levels;

FIGURE 18 is a graph showing the performance of a burner in accordance with the invention, calculated from chemical kinetics, as a function of the burner stoichiometry; and

FIGURE 19 is a graph comparing measured results operating a two stage burner in accordance with the invention in a biased firing mode with the fuel lean burner operating at 94% excess air and the fuel rich burner at 0.63 stoichiometry, maintaining an overall excess air level of 10%, with the same burners operating at the 10% excess air.

DESCRIPTION OF PREFERRED EMBODIMENTS

Figures 7 and 8 respectively depict a longitudinal cross-sectional and front end view of a two stage circular RMB 100 in accordance with the present invention. This arrangement employs separate windboxes 102 and 104 for the inner and outer portions of the burner. Air and FGR (recirculated flue gas) are provided under positive pressure by conventional fan means (not shown) via ducts 106 and 108 to both windboxes.

The air and flue gas mixture proceed through the inner burner throat 110 to the swirl vanes 112. The design of the swirl vanes and gas injectors correspond to the disclosure of my prior applications. At the leading edge of the swirl vanes, gas is injected in the same direction as the curvature of the swirl vanes, this arrangement being similar to that shown in Figures 3 through 6. The air, gas, flue gas mixture then passes through the swirl vanes resulting in a well mixed composition at the beginning of the quarl divergence 114. Ignition of the mixture occurs early in the quarl 116 and, at the axial position corresponding to the quarl exit, a significant amount of the fuel is combusted. The ignited gases proceed to a combustion chamber which in use is adjoined to the burner at the quarl exit.

The geometrical design of the inner burner is consistent with the design of the basic RMB -- see e.g. Figs. 3 to 6. The dimensions of the annular region defined by the ratio of the inner diameter of the swirl vanes divided by the outer diameter of the swirl vanes, is preferably in the range of 0.6 to 0.8. In addition, the product of the swirl number with the quarl outlet to inlet ratio is preferably in the range 1.0 to 3.0.

In order to help isolate the flame of the inner burner from the fluids in the outer portion of the burner, the quarl exit angle 118 would typically be zero degrees. However quarl exit angles ranging from either greater (diverging at the exit) or less than zero degrees (converging at the exit) may be desirable for some applications. To provide adequate residence time within the quarl for the inner burner, the quarl length/ quarl inlet diameter ratio should be a minimum of 1.75.

The air and flue gas mixture comprising the oxidant is also fed into the windbox 104 that supplies the outer burner. This oxidant stream is fed into the annular flow region or channel 120 between the outer burner wall 122 and the tube 124 extending back and partially defining the outer wall of the inner burner. The oxidant passes through two rows 126, 128 of gas injectors which extend radially into the outer burner annular flow channel 120. The gas injectors are fed fuel gas from manifold 131 into which the injectors extend and with which they communicate. Fuel gas to manifold 131 is provided via port 133. Wall 122 is secured to an outer refractory piece 135 by flange 139. Piece 135 essentially functions as a quarl for the outer burner. It has a central opening 137 forming part of flow channel 120.

Gas is fed through a number of injectors in rows 126, 128 which extend along radii. Each radial spud 132 has a series of injection holes which inject the gas normal to the oxidant flow in the same direction as the tangential component provided to the oxidant using the swirl vanes of the inner burner. However, fuel injection opposite to the swirl direction of the inner burner or in both directions simultaneously are also effective means of producing the desired mixing results. The totality of gas injection holes in effect define a grid of injection points, spaced by about 0.25 inches in the radial

direction and 0.5 inch in the circumferential direction. The objective is to provide premixed air/FGR/fuel before the outer burner gases are ignited by the combustion gases from the inner burner. The diameter of the holes are based on the rapid mix design disclosed in my prior said applications.

The outer burner gas spuds,shown in Figure 7, are aligned in two rows in order to generate additional mixing energy in the wake of each row. In the apparatus 100 there are two rows of spuds, each consisting of 20 cylindrical tubes. The tubes in one row are offset 15° from the tubes in the other row. The spuds may be aligned in either a single row or multiple rows. The spuds may take the shape of cylindrical tubes, oval tubes or other fabricated shapes having an minor outside diameter of approximately 0.25 inch. The cross-sectional area of each gas spud is typically at least 3 times the total area of the total injection holes in each spud, to provide uniform gas distribution to each hole.

As shown in Figures 9 and 10, swirl vanes 134 can be added to the outer annular or flow channel 120. The purpose of the swirl vanes 134 is to accelerate the mixing between the fuel and oxidant. The swirl vanes will also provide a degree of control over the flame shape with a higher swirl level resulting in a shorter, wider flame. Typically swirl vanes with an exit angle of 30 degrees are used, but vanes with exit angles in the range 10 to 50 degrees may be used to control the flame shape. The radial spuds 160 in the embodiment of Figures 9 and 10 are oval or flattened tubes, unlike the cylindrical tubes of Figures 7 and 8.

Within one outer tube diameter downstream of the gas injectors the outer burner flow will enter a refractory section. The refractory will extend downstream,

typically ending at the same axial position or extending slightly downstream, of the inner quarl. The refractory section could, however, be replaced with a cylinder formed from the surrounding water wall tubes, if sufficient space is not available in the water wall.

Figures 11 and 12 show a two stage RMB having a rectangular outer burner portion. This geometry corresponds to corner (or tangentially fired boilers) which make up a significant fraction of the large industrial and utility boiler market. Figure 13 also shows a view of 6 burners as they would appear in one corner of a typical corner fired boiler application. The inner burner is conceptually the same as the annular two stage burner described for Figures 7 through 10. The quarl of the inner burner has the same outside diameter as the smaller dimension of the rectangular boundary comprising the outer burner.

The gas injection manifold in the outer burner consists of a series of parallel vertical spuds 1/4 inch in width and spaced by one inch center to center. Parallel horizontal spuds would be equally effective in generating the desired rapid mixing. The cross-sectional area of each vertical injection spud is large enough to provide uniform gas distribution to each hole in the injector. Typically the cross-sectional area to each spud is at least 3 times the total area of the injection holes. Each spud has a series of holes spaced in 1/4 inch increments along its length. The gas injection spuds in the upper and lower burner cells are fed from separate manifolds located near to the upper and lower surface of the outer burner. The gas injection holes may be on either one side of the vertical manifold or on both sides depending on the application.

A screen, perforated plate or other mixing enhancer may be placed downstream of the gas injectors in the outer burner cells, to enhance mixing between the fuel and oxidant.

The objective of the gas distribution system and any screens or perforated plates, located downstream of the gas injection point, is to generate premixed fuel and oxidant upstream of the ignition point.

Experiments were conducted, with a burner having a geometry similar to that shown in Figure 11, in a 100 hp boiler where 4 MMBtu/hr represents full load. Tests were conducted varying the heat input (load) to the burner over the range 1.5 to 3.5 MMBtu/hr. Tests were also conducted varying the ratio of the heat input to the inner/total burner from 6.6% to 15%. Tests were conducted with both ambient combustion air and 500 F preheat.

The results of the tests varying the ratio of the inner/total burner heat input as a function of FGR rate with ambient air are shown in Figure 14. Burner stability and NO_x, at a constant FGR rate, are relatively unaffected by the ratio of the inner to total burner heat input. Figure 14 also compares the performance of the rectangular two stage RMB burner with the standard RMB. For FGR rates higher than about 20%, the two stage burner has lower NO_x emissions for a given FGR rate than the standard burner. Both the two stage burner and standard RMB are capable of NO_x emissions well below 10 ppm. Figure 15 shows the CO and total hydrocarbon emissions (THC) as a function of the FGR rate for the two stage burner. When NO_x emissions as low as 5 ppm were achieved, both the CO and THC emissions were below the detection limit of 1 ppm.

Figure 16 compares the effect of the inner/total burner heat input as a function of FGR rate for 500 F air preheat. Again the NO_x emissions and stability of the two stage burner were not a strong function of the ratio of the heat input ratio between the inner and total burner. For a given FGR rate above about 20%, the NO_x emissions of the two stage burner were lower than for the standard RMB for a given FGR rate.

The data in Figures 17 through 19 demonstrate that the two stage RMB has the capability of reducing NO_x emissions well below 10 ppm with FGR rates less than or equal to those used for the standard RMB. The low NO_x emissions can be maintained with less than 1 ppm CO or THC emissions.

Example

To illustrate the reduction in burner size which will result from a two stage design, the following example, comparing the burner diameters for a standard and two stage annular RMB, is given.

Design Criteria

- 100 MMBtu/hr maximum input
- 8 inches water pressure drop through burner at full load
- 500 F air preheat
- 500 F FGR temperature
- 15% excess air
- 20% FGR

Standard RMB

- Throat Diameter = 40 inches
- Quarl exit diameter (1.5 quarl expansion) = 60 inches

Two Stage RMB

- Inner Burner Quarl outside diameter = 20 inches (10 MMBtu/hr)
- Outer Burner Diameter = 33 inches

The two stage burner design will result in a maximum burner diameter of 33 inches compared to the standard RMB maximum diameter of 60 inches for the same burner capacity, FGR rate and pressure drop, with about the same flame stability, NO_x, CO and THC emissions. The size reduction occurs primarily for two reasons. First, since the outer burner does not require swirl vanes a higher axial velocity can be used for a given pressure drop. Second, since the flame in the outer burner is stabilized via the inner burner flame a quarl expansion for the outer burner is not required.

The two stage burner RMB can also be operated at high excess air levels to reduce NO_x levels down to extremely low levels in the same manner as the standard RMB. An example of the NO_x, CO and THC performance of the RMB as a function of the excess air level in shown in Figure 17. Excess air is equally effective as FGR in reducing NO_x levels down to below 3 ppm maintaining CO and THC emissions below 1 ppm.

Since the NO_x emissions can be controlled using the RMB equally effectively using excess air or FGR, a multi-burner RMB boiler can operate in what is commonly called a biased fired mode of operation to control NO_x emissions. Biased firing means, in a multi-burner furnace, that some burners operate air rich and others operate fuel rich. Figure 18 shows the performance of the RMB, calculated from chemical kinetics, as a function of the burner stoichiometry. The data in Figure 18 shows that even with air preheat, operating one burner near 80% excess

air and another burner at a stoichiometry of 0.6 should result in NO_x emissions from both burners less than 10 ppm.

Figure 19 compares the measured results operating a two burner RMB installation in a biased firing mode, with the fuel lean burner operating at 94% excess air and the fuel rich burner operating at 0.63 stoichiometry, maintaining an overall excess air level of 10% with the same burners both operating at the 10% excess air. The data in the figure 9 demonstrate that, without FGR, biased firing results in a reduction in NO_x emissions from 300 ppm to 20 ppm. If FGR is used biased firing reduces the amount of FGR required to achieve 10 ppm NO_x is reduced from 40% to less than 20%.

Although the data shown in Figure 19 is from a two burner standard RMB operation, the same performance would be expected from a multi-burner two stage RMB operation.

While the present invention has been particular set forth in terms of specific embodiments thereof, it will be understood in view of the present disclosure, that numerous variations on the invention are now enabled to those skilled in the art, which variations yet reside within the scope of the present teaching. Accordingly, the invention is to be broadly construed and limited only by the scope and spirit of the claims now appended hereto.

WHAT IS CLAIMED IS:

1. A forced draft burner apparatus for burning a gaseous fuel while producing low levels of NO_x, CO and hydrocarbon emissions; comprising:

a cylindrical inner burner having a tubular wall;

a generally cylindrical body mounted inside the tubular wall of the inner burner;

an annular flow channel being defined between said body and the inner wall of said tubular section, said channel constituting a throat for oxidant gases, and having a downstream outlet for the inner burner;

means for supplying oxidant gases to said throat of the inner burner;

a divergent quarl for said inner burner having its smaller end connected to said outlet of said inner burner, and exiting into a combustion chamber;

a plurality of curved axial swirl vanes being mounted in said annular flow channel of the inner burner to impart swirl to said oxidant gases flowing downstream in said throat;

inner burner fuel gas injection means for the inner burner being provided in said annular channel proximate to said swirl vanes for injecting said gas into the flow of oxidant gases at a point upstream of said outlet end in a manner that results in uniform mixing of the fuel and oxidant upstream of the ignition point;

said fuel gas injection means for said inner burner comprising a plurality of spaced gas injectors each being defined by a gas injection hole and means to feed the gas thereto; the ratio of the number of gas injection holes to the transverse cross-sectional area of the annular flow channel of the inner burner being at least 200/ft^2;

an outer burner surrounding at least a portion of said inner burner and including a wall spaced from the outer wall of the inner burner to define an outer burner flow channel having a downstream outlet end for gases provided to said channel;

means for providing a flow of oxidant into the outer burner flow channel; and

outer burner fuel gas injection means for the outer burner being provided in said outer burner flow channel, upstream of the outer burner outlet end, comprising a plurality of spaced gas injectors, each being defined by a gas ejection hole and means to feed the gas thereto; the ratio of the number of gas injection holes to the transverse cross-sectional area of the flow channel of said outer burner being at least 200/ft^2.

2. Apparatus in accordance with claim 1, wherein said outer burner is of rectangular cross-section.

3. Apparatus in accordance with claim 1, wherein said outer burner is of cylindrical cross-section.

4. Apparatus in accordance with claim 1, wherein said outer burner is of irregular cross-section.

5. Apparatus in accordance with claim 1, wherein a means are provided to premix recirculated flue gases into the combustion air for both the inner and outer burners.

6. Apparatus in accordance with claim 1, where the heat input ratio of the inner/outer burner at full load is in the range of 5% to 20%.

7. Apparatus in accordance with claim 1, wherein the product of the swirl number and quarl inlet to outlet number for the inner burner is in the range 1.0 to 3.0.

8. Apparatus in accordance with claim 1, wherein the ratio of the inner diameter to the outer diameter of the inner burner swirl vanes is in the range 0.6 to 0.8.

9. Apparatus in accordance with claim 1, wherein turbulence enhancing means are provided in the outer burner to promote fine scale turbulence into the flow and generate microscale mixing between the fuel and oxidant prior to combustion.

10. Apparatus in accordance with claim 8, wherein swirl vanes are provided in the outer burner flow channel to promote mixing and control flame shape.

11. Apparatus in accordance with claim 8, wherein screens or perforated plates are provided in said annular flow channel to promote microscale mixing.

12. Apparatus in accordance with claim 1, having an inner burner quarl length to inlet diameter ratio of 1.75 or greater, with a quarl outlet to inlet diameter ratio of approximately 1.5.

13. Apparatus in accordance with claim 1, in which said inner burner gas injectors are located at the leading edge of said swirl vanes and inject said fuel gas counter-current, co-current or both counter-current and co-current to the direction of the tangential component

of the flow imparted by the swirl vanes of the inner burner.

14. Apparatus in accordance with claim 1, in which said inner burner gas injectors are located at the trailing edge of said swirl vanes and inject said fuel gas counter-current, co-current or both counter-current and co-current to the direction of the tangential component of the flow imparted by the swirl vanes of the inner burner.

15. Apparatus in accordance with claim 1, in which inner burner gas injectors are disposed on a plurality of hollow concentric rings which are mounted in said throat proximate to the swirl vanes of the inner burner.

16. Apparatus in accordance with claim 15, comprising at least two spaced rings, the holes on the outer ring facing toward the axis of said conduit and the holes on the inner ring facing away from said axis, whereby to produce a flow of gas from each ring toward each other.

17. Apparatus in accordance with claim 1, wherein said outer burner fuel gas injection means comprise gas spuds along the radii of the outer burner with injection holes being oriented to eject fuel gas counter-current, co-current or both counter-current and co-current to the direction of swirl of the inner burner.

18. Apparatus in accordance with claim 1, wherein said outer burner fuel gas injection means comprise multiple rows of gas spuds, at more than one axial position, along the radii of an annular outer burner with injection holes being oriented to eject fuel gas counter-current, co-current, or both counter-current and co-current to the direction of swirl of the inner burner.

19. Apparatus in accordance with claim 1, having gas injectors parallel to one or more sides of a rectangular outer burner.

20. Apparatus in accordance with claim 1, wherein said generally cylindrical body spaced inwardly from said tubular section includes a liquid fuel injector means, which thereby provides the burner both gas and liquid fuel firing capabilities.

21. Apparatus in accordance with claim 20, wherein said liquid injector comprises a liquid feed tube extending along the axis of said generally cylindrical body; and a nozzle from the end of said tube extending from the outlet end of said throat, a hollow cylinder surrounding said tube and being open at the end toward said nozzle; and said apparatus including means for diverting air from said windbox to said hollow cylinder to provide an air stream preventing coke and ash particles from depositing on the liquid gun during liquid firing.

22. Apparatus in accordance with claim 1, wherein the geometry of one or more of the outer burners corresponds to the standard burner openings of a corner fired boiler.

23. A method of operating a multi-burner two stage RMB furnace in a biased firing mode, wherein some burners are operated fuel lean and other burners are operated fuel rich, resulting in an overall excess air level similar to that present when all burners are operating at the same stoichiometry.

24. A method in accordance with claim 23, wherein FGR is added to both burners.

25. A method in accordance with claim 23, wherein FGR is added to the fuel rich burners only.

FIG. 1

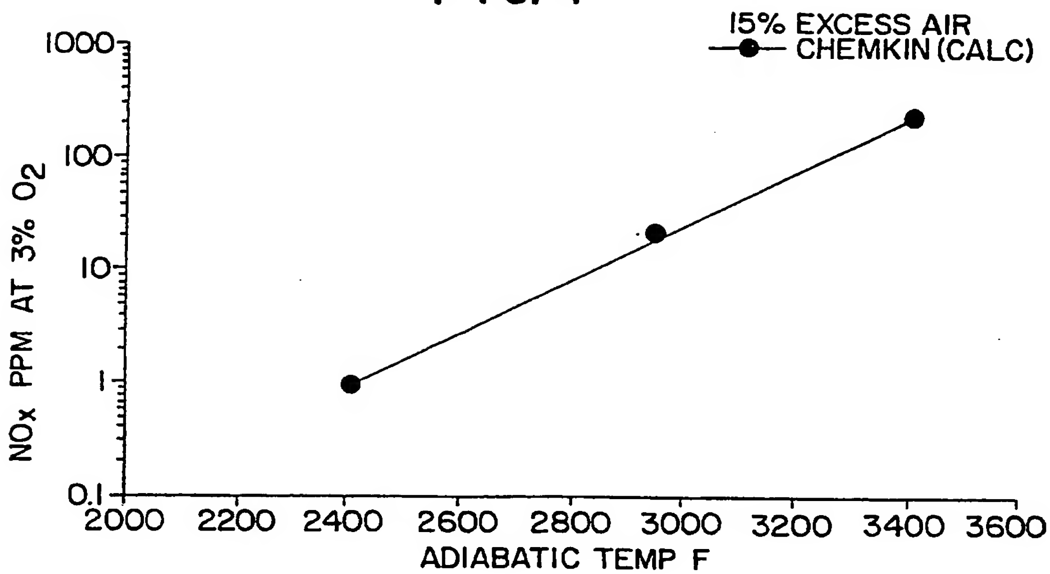

15% EXCESS AIR
CHEMKIN (CALC)
1000
100
10
1
0.1
NOx PPM AT 3% O2
2000 2200 2400 2600 2800 3000 3200 3400 3600
ADIABATIC TEMP F

FIG. 2

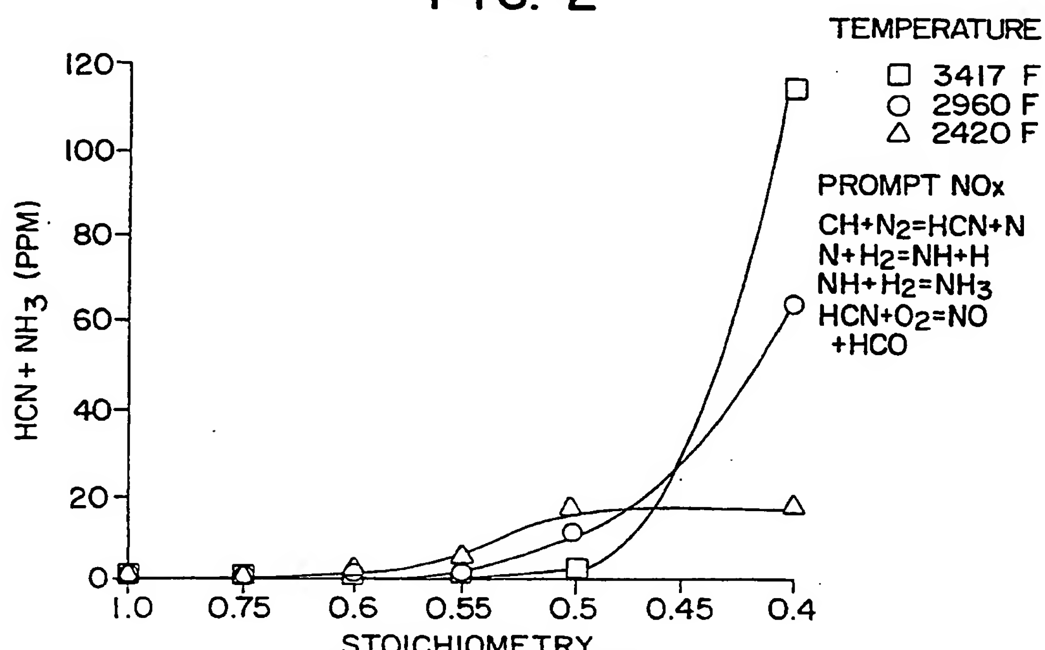

TEMPERATURE
3417 F
2960 F
2420 F
PROMPT NOx
CH+N2=HCN+N
N+H2=NH+H
NH+H2=NH3
HCN+O2=NO
+HCO
120
100
80
60
40
20
0
HCN + NH3 (PPM)
1.0 0.75 0.6 0.55 0.5 0.45 0.4
STOICHIOMETRY

FIG. 3

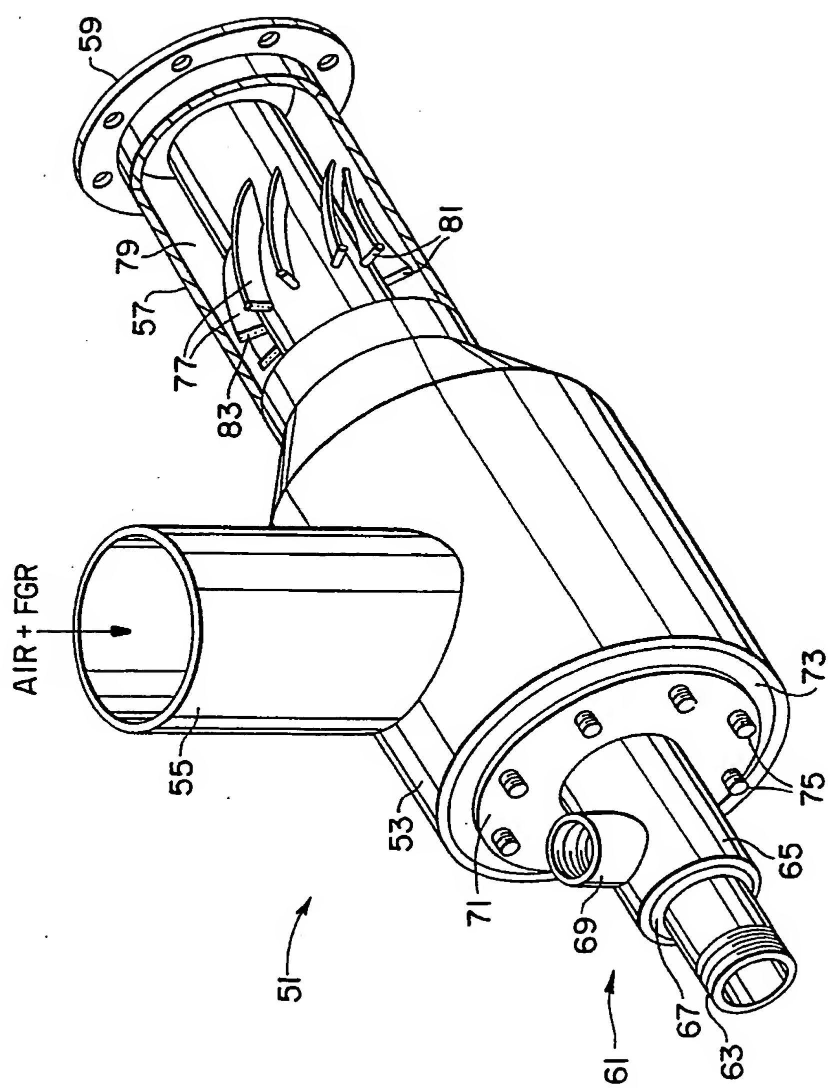


FIG. 5

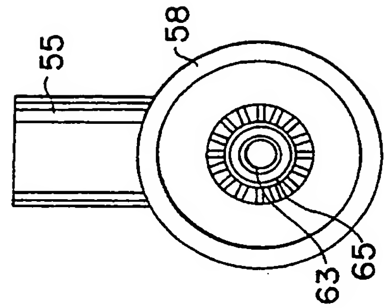


FIG. 4

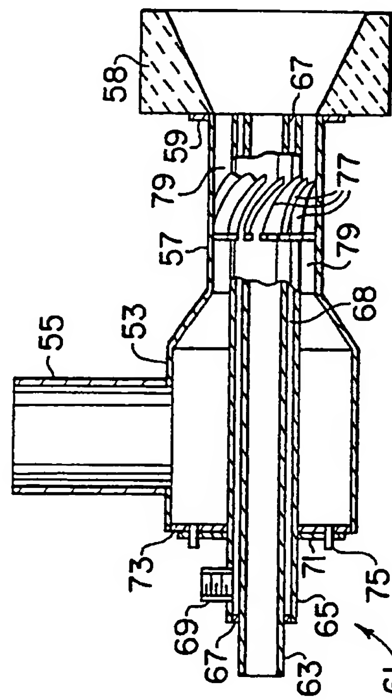
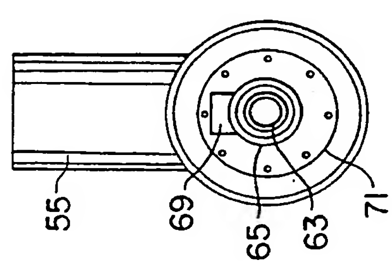


FIG. 6



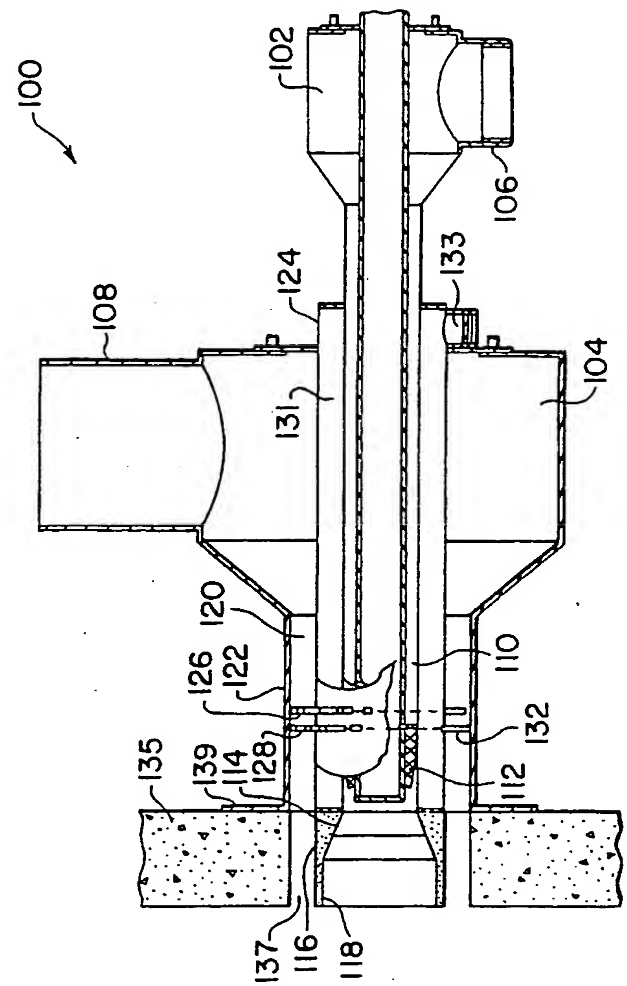

100
102
106
108
124
133
104
131
120
110
126
122
132
135
139
114
128
112
137
116
118

FIG. 7

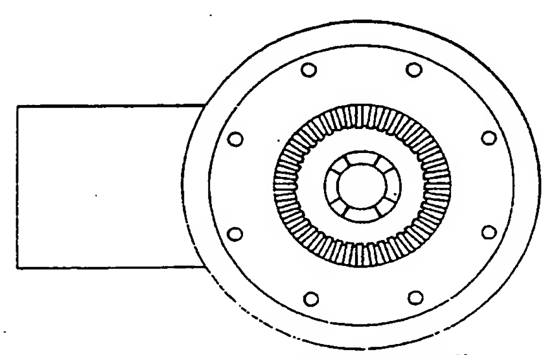


FIG. 8

FIG. 9

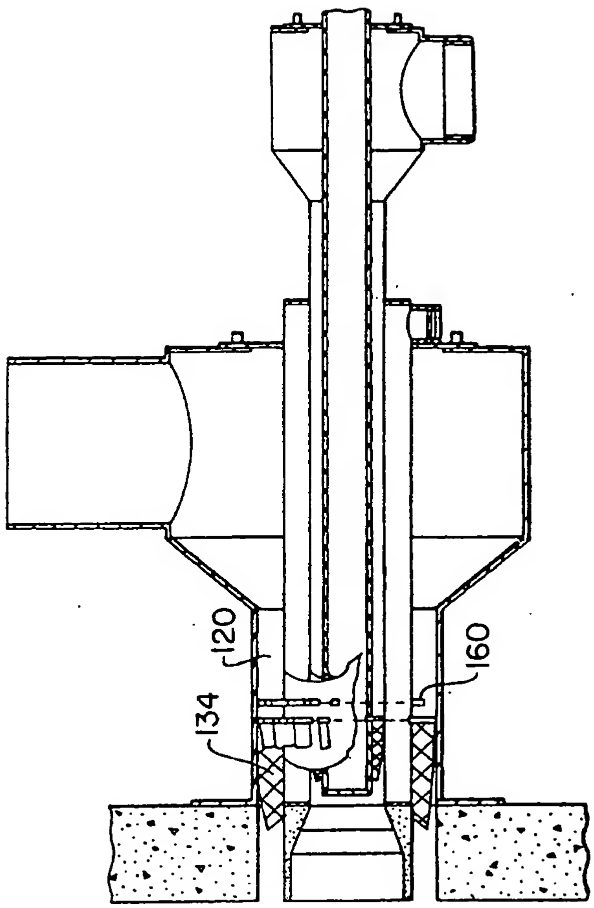
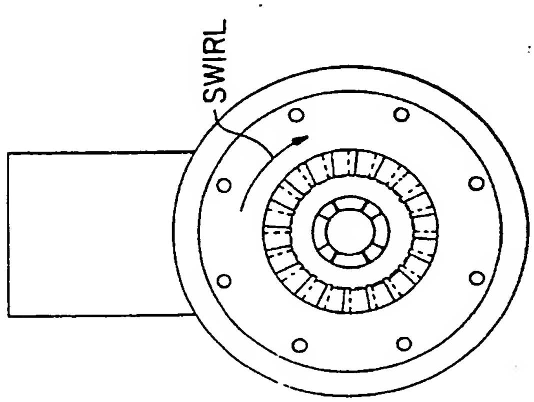


FIG. 10



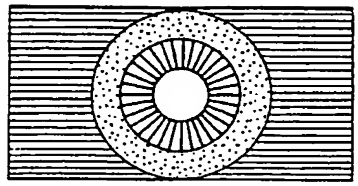


FIG. 12

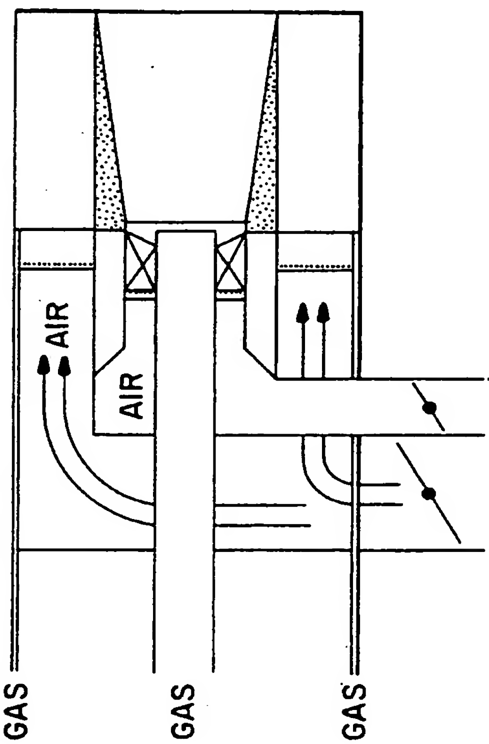


FIG. 11

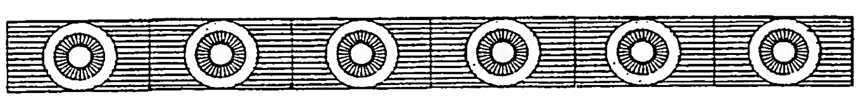


FIG. 13

FIG. 14

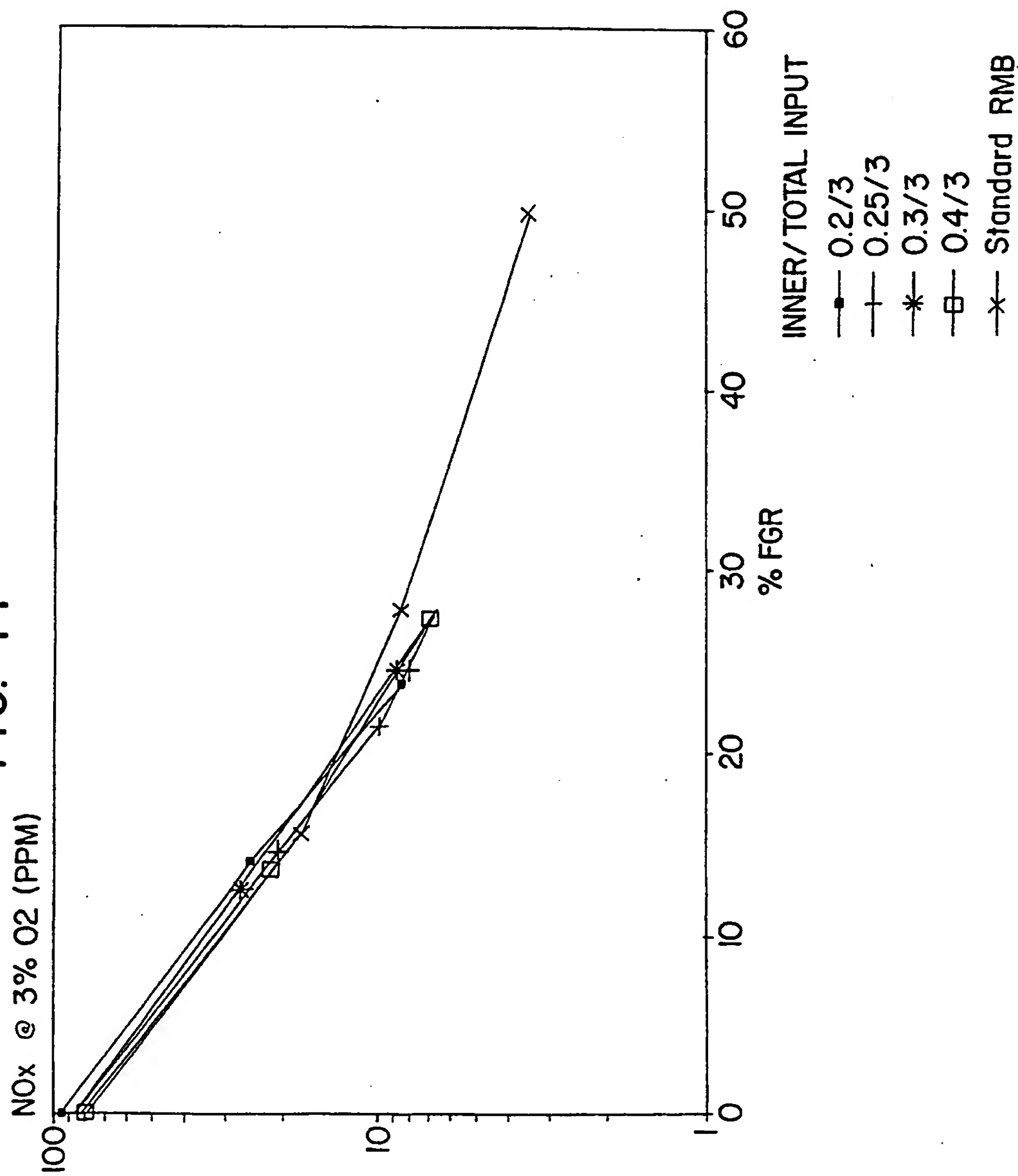
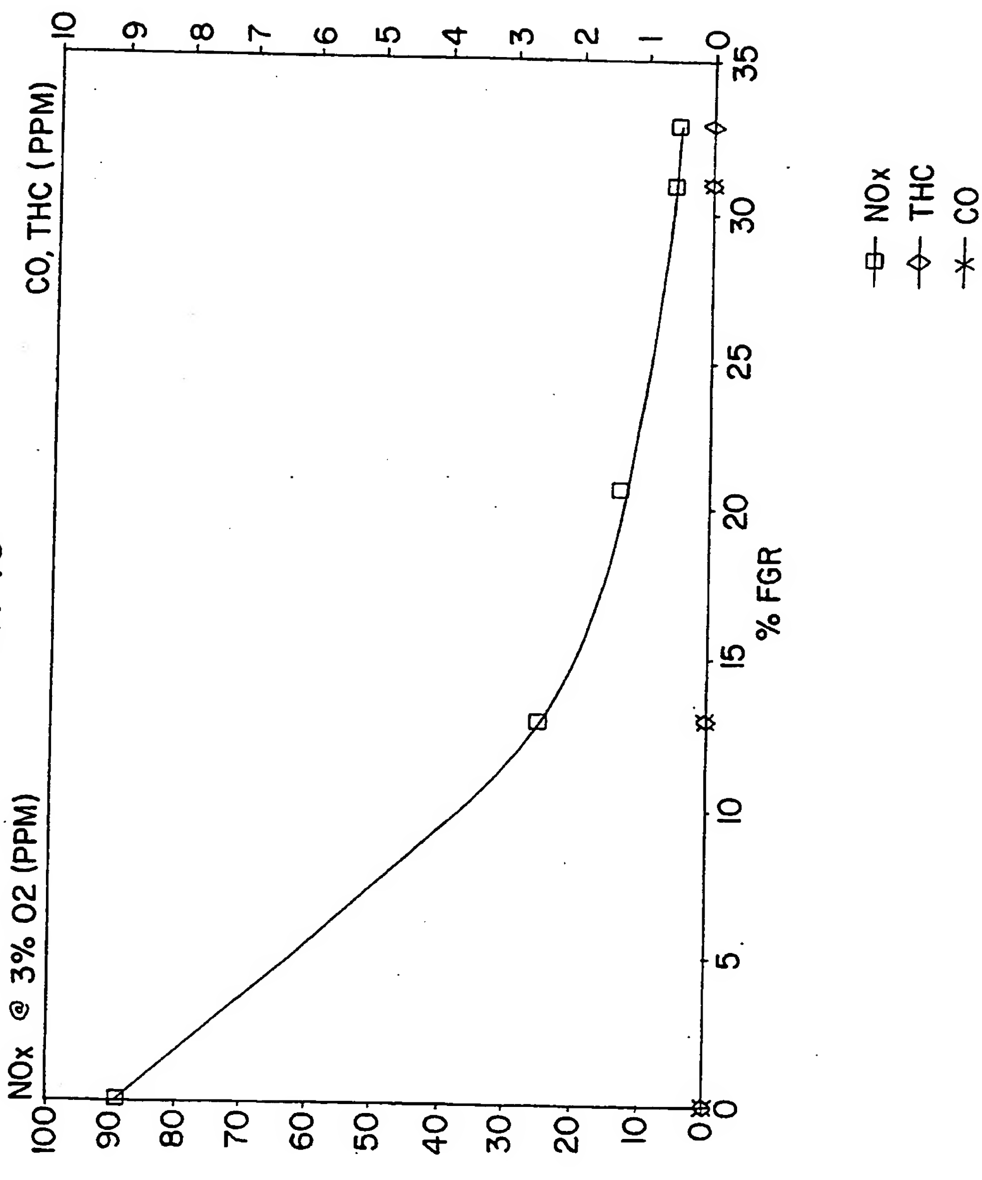
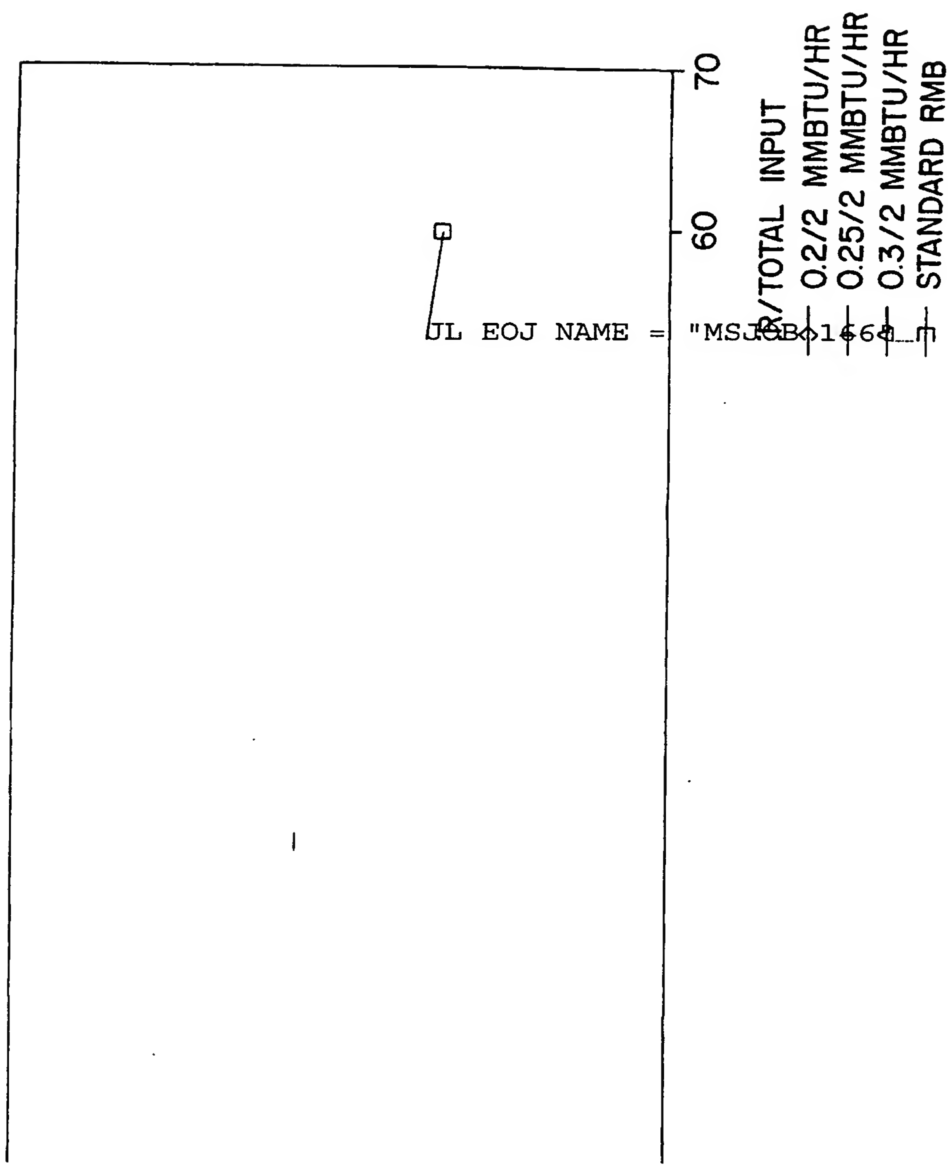

NOx @ 3% O2 (PPM)
100
10
1
0
10
20
30
40
50
60
% FGR
INNER/TOTAL INPUT
0.2/3
0.25/3
0.3/3
0.4/3
Standard RMB

FIG. 15


NOx @ 3% O2 (PPM)
CO, THC (PPM)
% FGR
0
5
10
15
20
25
30
35
100
90
80
70
60
50
40
30
20
10
0
10
9
8
7
6
5
4
3
2
1
0
NOx
THC
CO


70
60
R/TOTAL INPUT
0.2/2 MMBTU/HR
0.25/2 MMBTU/HR
0.3/2 MMBTU/HR
STANDARD RMB